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Design and Evaluation of High Contract Ratio Gearing

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16. Abstract

Scoring test and surface fatigue tests were conducted on spur gears of a standard design and spur gears of a high contact ratio design. All testing was conducted in the Sikorsky 6-inch dynamic test facility at a speed of 8000 rpm using 23699 oil. Bulk gear temperatures were also recorded. The surface fatigue tests were run at Hertz stresses of 165×10^7 N/m^2 and 145×10^7 N/m^2 (240,000 and 210,000 psi), respectively.

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ABSTRACT

Scoring tests and surface fatigue tests were conducted on spur gears of a standard design and spur gears of a high contact ratio design. All testing was conducted in the Sikorsky 6-inch dynamic test facility at a speed of 8000 rpm using 23699 oil. The scoring tests were run in a step load fashion until scoring was observed. Bulk gear temperatures were also recorded. The surface fatigue tests were run at Hertz stresses of 165 X 10^7 N/m^2 and 145 x 10^7 N/m^2 (240,000 and 210,000 psi) respectively.

The HCR gears scored at an average gear bulk temperature of 123°C (253°F). The corresponding gear scoring index was 461 degrees.

The 10-percent life of the HCR gear design was approximately two times the life of the standard gear design at the same load level.

INTRODUCTION

Among the goals of current aerospace research and development programs are increased reliability and power-to-weight ratio of power transmission systems. Consequently, considerable effort has been directed, in the last decade, at improving the performance of spur gears with respect to these two criteria. This effort has been primarily aimed at the development of new gear materials; such as Vasco-X2, Super Nitralloy, M-50, CBS600 and EX00053; and advanced manufacturing methods, such as high energy rate forgings and roll-formed gears. Although some research has been done on alternate gear tooth forms such as the Wilhaber/Novikov conformal system, the profile geometry of aerospace gearing has remained basically unchanged over the years.

One possibility for improving the performance of spur gears, that has not been fully explored, is increased contact ratio. Present day spur gearing is usually designed to operate at contact ratios between 1.4 and 1.6, where contact ratio is generically defined as the average number of tooth pairs in contact under static load conditions. Thus, a typical contact ratio of 1.5 means that, ideally, two tooth pairs are in contact half the time and only one tooth pair is in contact the other half. High contact ratio gearing (HCRG) is usually defined as gear meshes that have at least two tooth pairs in contact at all times, i.e., contact ratios of 2.0 or more. Because the transmitted load is always theoretically shared by at least two pairs of teeth in this configuration, the individual tooth loading is considerably less for HCRG than for present low contact ratio designs, thereby potentially decreasing gear tooth stresses. HCRG, however, inherently requires gear teeth with lower pressure angles, finer pitches, or increased working depths; all of which tend to increase the tooth bending stress per individually applied tooth load. In addition, it would be expected that HCRG would be more sensitive to tooth spacing errors and profile modifications because of the multiple tooth contacts and attendent load sharing requirement. The basic problem to be resolved in this study is whether the lower tooth loads occurring in the high contact ratio design configuration more than offset the effects of the weakened tooth form especially when run under dynamic load conditions.

In the program reported on herein, the prime objective was to design, fabricate, and dynamically test a high contact ratio gear set in the Sikorsky gear tester. The test results are compared to those of a standard low contact ratio design fabricated from the same material and tested under identical conditions in the same tester.

APPARATUS, SPECIMENS, AND PROCEDURE

Test Specimens

Both the high contact ratio test gears and the standard test gears were manufactured from a single lot of vacuum melt AMS 6265 (A1S1 9310) gear steel. The gears were machined from conventional pancake forgings (Figure 1) and serialized as shown in Table 1.

Dimensions for the test gears are shown in Figures 2 and 3. The high contact ratio configuration has a contact ratio of 2.20 whereas the standard design has a contact ratio of 1.62. The case hardness of the gears was Rockwell C 60 to 64. The core hardness was Rockwell C 30 to 45. All gears had a nominal surface finish on the contacting tooth face of .330 micrometers (13 μ in). The involute profile modification at the tip of the high contact ratio gear tooth was a second order curve starting at 90 percent of the addenda with a max value at the tip of .0406 mm (.0016 in), see Table 2. The standard gear tooth had a max tip relief of .0508 mm (.0020 in). which started at the highest point of single pair contact with an undefined curve shape.

Analysis

The analysis of the gear teeth was accomplished with the aid of the Hamilton Standard dynamic analysis gear program, modified under this contact to take into account high contact ratios and nonstandard tooth forms. While no attempt will be made here to describe the computer program in detail, it will be instructive to examine the analytical model on which it is based. A free body diagram of a pair of meshing high contact ratio gears is shown in Figure 4. With the assumptions that frictional forces are negligible and that input and output torques T_1 and T_2 are constant and equal to their average values, the dynamic model for a meshing pair of gears can be idealized as shown in Figure 5. This figure represents a set of high contact ratio gears with three tooth pairs in contact. The mass "m" represents the effective mass of the gear set acting along the line of action. It is being acted upon by the average transmitted load, W_n, the spring forces of tooth pairs n-1, n, and n+1; and by the force resulting from the effective damping "b". The model, with a length equal to the zone of the tooth contact, moves along the line of action with average velocity, V. The various tooth pairs come into engagement at the engagement cam, move along the line of action, and then out of mesh at the disengagement cam. The engagement and disengagement cams, which represent the gear tooth profile tip reliefs, are considered to be constant acceleration cams which simulate the dynamic action of the gear teeth as they roll into and out of mesh. Each tooth pair representation in Figure 5 is shown with a positive tooth error. These errors serve to increase the loads on the effective mass due to the premature meshing of the tooth pairs.

Table 1. Gear Identification and Serialization

Forging Part Number	Gear Part Number	Serial Number
38017-01000-002	38017-01000-105	B105-00001-24
38017-01000-002	38009-00063-	B063-00001-52

Table 2. High Contact Ratio Gear Tip Modification

Roll Angle	Modification, mm	(in.)
29° 6.17'	-.0356/-.0406	(-.0014/-.0016)
28° 57.3'	-.0203/-.0254	(-.0008/-.0010)
28° 48.2'	-.0102/-.0152	(-.0004/-.0006)
28° 39.2	-.0051/-.0076	(-.0002/-.0003)
28° 30.2	-.0000/-.0025	(-.0000/-.0001)

Referring to Figure 5, the differential equation of motion between the two gears can be written as:

$$M \ddot{S}_r + d \dot{S}_r + \sum_{j=1}^{n} W_j = W_N \tag{1}$$

where W_j is the individual tooth load, W_N is the total applied load, $S_r = S_2 - S_1$ the relative motion between the two gears, d is the damping factor, and n is the number of tooth pairs in contact. This equation in dimensionless form is:

$$\ddot{y} + 2\xi\dot{y} + \sum_{j=1}^{n} W_j/W_N = 1 \tag{2}$$

where

$$y = S_r \, k_0/W_N \qquad k_0 = \text{stiffness}$$

$$\dot{y} = dy/\omega dt \qquad \omega = \sqrt{k_0/M}$$

and ξ is the damping coefficient.

Because of the variable tooth pair stiffnesses acting during the mesh, differential equation (2) is non-linear so that the closed form solutions will apply for only an instant in time and are thus piecewise continuous through the mesh. To solve this equation a time history solution is used, stepping through the meshcycle in small dimensionless time increments during which the tooth stiffness is assumed to be constant. In this way solutions can be obtained for the deflections and resulting dynamic tooth loads. Implicit in this procedure however, is the knowledge of the initial values of y and $\dot{y}$. These can be obtained by interaction on the basis that after the passage of one gear mesh ($\Delta S = 1$), the gear displacement and velocity must be the same as that for the starting condition, when no gear errors are involved. The solution for the case with tooth errors is done then in two steps; first, the iterative solution is obtained for the zero-error case as discussed above. Then starting with the known initial condi tions of y and $\dot{y}$, the analysis is run on a consecutive mesh basis, introducing the specific error on any tooth or teeth desired. A more complete discussion of the details of this procedure is contained in References (1), (2), and (3).

The results of this dynamic analysis for a tooth load of 11,120 N (2500 lb) is shown in Table 3 for both the standard and high contact ratio designs.

Table 3. Six-Inch Diameter Test Gear Data

FEATURE	CONVENTIONAL	HCR
Diametral pitch	8	8
Pressure Angle, deg	22.5	20
Circ. Tooth Thickness cm (in)	.4935/.4910 (.1943/.1933)	.4935/.4910 (.1943/.1933)
Top Land, cm (in)	.206 (.081)	.147 (.058)
Contact Ratio	1.628	2.200
Tangential Tooth Load, N (lb)	11120 (2500)	11120 (2500)
Bending Stress, N/m^2 (psi)	$.810 \times 10^9$ (117,600)	$.635 \times 10^9$ (92200)
Hertz Stress, N/m^2 (psi)	1.65×10^9 (239,400)	1.45×10^9 (209,800)

Gear Fabrication

The test gears were manufactured by Precision Gear Incorporated of Corona, New York to Sikorsky requirements. The fabrication of the test gears was in accordance with the manufacturing schedule shown in Table 4.

The forgings, when received at Precision Gear, were serialized and normalized to Rc 24-29 before rough machining. The forgings were rough-machined, copper plated, and then remachined to remove the copper plate from the areas to be carburized. The gears were then deburred, carburized and heat treated using established and varified procedures for the A1S1 9310 material. Final grinding of the test gears to the blueprint requirements was accomplished on a Reishauer machine with special attention paid to tooth spacing and finish requirements. The final gear configuration is shown in Figures 6 and 7. An involute profile trace of the two test gears is shown in Figure 8 and a typical tooth spacing chart is shown in Figure 9.

Table 4. Six-Inch Test Gear Manufacturing Schedule.

Step	Process	Purpose	Lot Size	Control
1	Serialize Forgings	Identification	Individually	Serialize upon Receipt.
2	Rough machine	Remove excess stock	Individually	Random order
3	Normalize to R_C 24-29	Machineability & structural refinement	All together	
4	Pre-copper plate machining	Mask areas for carburizing	Individually	Random order
5	Copper plate		All together	
6	Machine areas to be carburized	Remove copper plate	Individually	Random order
7	Deburr	Remove sharp edges	Individually	Random order
8	Carburize & heat treat	Heat-treat cycle	All together	
9	Strip copper plate	Remove copper finish	All together	
10	Remachine (grind)	Establish final dimensions	Individually	Random order
11	Machine I.D. & cut keyway	Establish bore and keyway dim.	Individually	Random order
12	Axial grind teeth	Establish final tooth dimensions	Individually	Random order
13	Stress relieve & nital etch	Inspection for grinding burns		Random order 100%
14	Magna flux	Inspect for grinding cracks		100% inspection

Test Facility

The gear test facility used in this series of tests was a "four square" regenerative test stand driven by a 150-HP electric motor. Two pairs of 6-inch-pitch-diameter test gears, external to the main gearbox, were tested simultaneously. The test stand is illustrated pictorially and schematically in Figures 10, 11, and 12. Torque loading of the gear train was achieved by applying an axial load via a hydraulic cylinder to a pair of sliding helical gears in the main gearbox. Since torque is locked into the system through the helical gear set, the drive power requirement was only that required to overcome the gearbox friction. Rotational speed was accomplished with a constant-speed electric motor and an electric clutch. The output of the clutch was controllable and allowed operation of the test stand at any speed from 0 to 8000 rpm.

Separate lubrication oil systems consisting of reservoir, pump, filter and heat exchanger were used for each set of test gears and the main gearbox.

The test gears were installed into two separate gear meshes with individual lubrication systems separate from the facility lube system. Failure detection devices were incorporated into each of the gearbox sections. Test functions were monitored as follows:

1. A broken tooth detector automatically stopped the test when a test gear tooth broke.
2. A chip detector, in conjunction with an accelerometer, detected compression type (spalling, pitting) failures of the test gear teeth.
3. Thermo-couples continuously monitored the bearing temperatures of the facility.
4. A lube oil flow measuring system monitored the flow to each test gear set.

An operator's console, Figure 13, located in an adjacent control room permitted operation of the test stand in a non-hazardous area. The operator's console included all the required controls for operating and monitoring the test.

EVALUATION TESTS

Dynamic testing consisted of a scoring test on the high contact ratio gears and a constant-load fatigue test on both configurations to compare the relative lives of the high contact ratio design and the standard tooth design.

Initial checkout of the test facility included a dynamic torque calibration. A thin-walled shaft extension was fabricated, strain gaged, and statically calibrated. Test gears were installed and a high-speed slip ring was mounted outboard of the test gear set. The dynamic calibration was accomplished at various loads and speeds. The lubricating oil MIL-L-23699 was used for both gear sets during the torque calibration and all subsequent tests.

Scoring Tests

Eight scoring tests were conducted on the high contact ratio gears at a speed of 63.5 m/sec (12,500 ft/min.). The test gears were step loaded in increasing increments at test intervals of 20 minutes and the teeth examined for scoring at the end of each test interval without removing the gears from the test rig. These tests were run at reduced oil flow to induce scoring and establish baseline conditions for comparative testing. The test conditions were .013 ℓ/sec (.2 gpm) oil flow and an oil-in temperature of 65.5°C (150°F). Bulk gear temperature was measured at the end of each interval. Testing was continued until a scoring threshold was reached. Scoring test results are shown in Table 5.

Table 5. Summary of Scoring Test Results

S/N	Load @ Scoring N (lb)	Bulk Temp. °C (°F)	Scoring Index
03/25	8771 (1972)	115 (240)	450
09/17	8850 (1990)	112 (233)	452
16/45	8833 (1986)	115 (240)	452
50/25	8850 (1990)	121 (250)	452
17/33	9216 (2072)	123 (254)	461
32/49	9500 (2136)	125 (258)	469
3/33	9518 (2140)	128 (258)	469
49/34	10050 (2260)	135 (276)	482

Fatigue Life Tests

These dynamic tests consisted of evaluating the relative pitting lives for the two design configurations at one load level.

Initial operation of any test gear pair consisted of a break-in run that was conducted in the following manner. Speed was increased slowly to about 3000 rpm at zero load. After a satisfactory checkout of all instrumentation, 25 percent of full load was slowly applied. After 10 minutes, speed was slowly increased to 8000 rpm (100 percent speed). A 5-minute time period at this condition was followed by slowly increasing the load to 75 percent. After a satisfactory instrumentation checkout, the load was increased to 100 percent.

Testing was continued until either a failure occurred or 1.2×10^8 cycles (250 hours) were completed. In the event that only one gear of a pair failed, both gears were replaced. Gear pairs were randomly selected.

The lubricant inlet temperature was held constant at 65.5°C (150°F) and was initially directed into and out of mesh at a flow rate of .025 ℓ/sec (.39 gpm). The flow rate was subsequently increased, by increased pressure, to .050 ℓ/sec (.78 gpm) and the jets realigned to direct the oil radially into mesh to reduce gear bulk temperature and prevent scoring and breakage.

The gears were tested at 11120 N (2500 lbs) tangential tooth load. The test results are shown in Table 6.

Table 6. Summary of Pitting Test Results

Run No.	Gear S/N Side No.1 L.H.	Gear S/N Side No.1 R.H.	Gear S/N Side No.2 L.H.	Gear S/N Side No.2 R.H.	Load N (Lbs)	Cycles X 10^{-6}	Notes
1	17 STD	11 STD	02 STD	09 STD	11120 (2500)	10.59	S/N 17/11 Failed
2	19 HCR	46 HCR	02 STD	09 STD	"	31.02	S/N 19/46 Failed
3	09 STD	02 STD	12 HCR	10 HCR	"	.84	S/N 10 Failed in Bending
4	09 STD	02 STD	22 HCR	40 HCR	"	43.89	S/N 09/02 Failed
5	16 STD	12 STD	48 HCR	38 HCR	"	.080	S/N 48 Failed in Bending
6	16 STD	12 STD	44 HCR	14 HCR	"	.040	S/N 16212 Scored
7	39 HCR	18 HCR	14 STD	10 STD	"	.040	S/N 39818 Scored
8	03 STD	07 STD	14 STD	10 STD	"	.112	S/N 03/07 Failed
9	18 STD	06 STD	14 STD	10 STD	"	80.70	S/N 18 Failed Pitting & Bending
10	21 STD	08 STD	14 STD	10 STD	"	132.2 42.27	S/N 14/10 Failed S/N 21/08 Failed in Bending
11	28 HCR	04 HCR	23 HCR	26 HCR	"	.025	S/N 28 Failed in Bending
12	43 HCR	15 HCR	42 HCR	24 HCR	"	.024	S/N 43 Failed in Bending
13	13 STD	23 STD	15 STD	20 STD	"	.88	S/N 15 Failed in Bending
14	13 STD	23 STD	05 STD	04 STD	"	25.05	S/N 05/04 Failed

Table 6. Summary of Pitting Test Results (Cont'd)

Run No.	Gear Side S/N No.1 L.H.	Gear Side S/N No.1 R.H.	Gear Side S/N No.2 L.H.	Gear Side S/N No.2 R.H.	Load N (Lbs)	Cycles X 10^{-6}	Notes
15	13 STD	23 STD	31 HCR	21 HCR	11120 (2500)	39.02	S/N 13/23 Failed
	Improvements made to oil jets						
16	02 HCR	27 HCR	31 HCR	21 HCR	"	64.08	S/N 21 Failed by Pitting & Bending
17	41 HCR	36 HCR	06 HCR	13 HCR	"	2.93	S/N 36 Failed in Bending
18	08 HCR	11 HCR	06 HCR	13 HCR	"	77.66	S/N 06/13 Failed
19	08 HCR	11 HCR	47 HCR	35 HCR	"	92.74	S/N 08/11 Failed
20	20 HCR	01 HCR	47 HCR	35 HCR	"	2.64	S/N 01 Failed Bending
21	52 HCR	05 HCR	47 HCR	35 HCR	"	24.96	S/N 47/35 Failed
22	27 HCR	42 HCR	24 HCR	22 HCR	"	.161	S/N 42 Failed in Bending
23	13 HCR	41 HCR	24 HCR	22 HCR	"	123.7	S/N 24/22 Runout
24	38 HCR	20 HCR	12 HCR	44 HCR	"	97.44	S/N 38 Failed in Bending
25	23 HCR	37 HCR	12 HCR	44 HCR	"	80.16	S/N 23/37 Failed
26	08 HCR	33 HCR	12 HCR	44 HCR	"	177.6	S/N 12/44 Runout

Table 6. Summary of Pitting Test Results (Cont'd)

Run No.	Gear S/N Side No.1 L.H.	Gear S/N Side No.1 R.H.	Gear S/N Side No.2 L.H.	Gear S/N Side No.2 R.H.	Load N (Lbs)	Cycles X 10^{-6}	Notes
27	08 HCR	33 HCR	06 HCR	03 HCR	11120 (2500)	152.2	S/N 08/33 Runout
28	39 HCR	45 HCR	31 HCR	07 HCR	"	4.99	S/N 45 Failed in Bending
29	09 HCR	04 HCR	31 HCR	07 HCR	"	.388	S/N 09 Failed in Bending
30	21 STD	03 STD	31 HCR	07 HCR	"	98.75	S/N 21/3 Runout
31	21 STD	03 STD	31 HCR	07 HCR	"	111.4	S/N 31/7 Runout
32	32 STD	19 STD	10 STD	16 STD	"	74.7	S/N 23/19 Failed
33	04 STD	14 STD	10 STD	16 STD	"	187.7	S/N 04/14 Runout
34	04 STD	14 STD	10 STD	16 STD	"	262.4	S/N 10/16 Runout

TEST RESULTS AND DISCUSSION

Results of Evaluation Tests

Scoring Tests

The results of the scoring tests are tabulated in Table 5 and are shown plotted in Figure 14. The calculated AGMA Flash Temperature Scoring Index with a fixed friction coefficient of .06, is shown in Table 5 along with the measured bulk temperature at the scoring load.

Pitting Tests

The pitting test results are summarized in Tabular form in Table 6 and in the form of comparative Weibull plots in Figure 15. The Weibull curves were derived using linear regression analysis to determine the best fit line representing the data. Significant bending failure points were considered as suspended data in the analysis. The 10-percent failure life was 39 X 10^6 cycles for the HCR gears and 19 X 10^6 cycles for the standard gears. The HCR gears tested have approximately 2 times the 10-percent life of the standard gears at equal loads.

Discussion of Results

Scoring Tests

The test results indicate a mean scoring load of 9800 N/cm (5600 lb/in). The average gear bulk temperature at the scoring load was 123°C (253°F). This result agrees favorably with the results of Reference 4. The corresponding calculated AGMA Flash Temperature Scoring Index was 460 degrees. The AGMA scoring criteria of Reference 5 indicates a scoring probability of 90 percent at this temperature.

Pitting Tests

The maximum calculated Hertz stress for the standard and HCR gears were 165 X 10^8 N/m^2 and 145 X 10^7 N/m^2 (240,000 and 210,000 psi) respectively. The test gears showed a mixed mode of failure before increased lube flow (bending, pitting, and scoring). This was not entirely unexpected since the gears were designed using a conventional balance between bending and pitting stresses and the gears were tested with full face contact. These conditions are more representative of actual aircraft design procedures. Another reason for selecting the standard configuration is that this design was used in an earlier test program and some baseline data was already available. Any attempt to bias the HCR design to preclude bending failures would have been futile since any strengthening would have the effect of reduced contact ratio.

Early in the test program some of the gears showed evidence of scoring and excessive heat leading to premature tooth breakage. Scoring is unacceptable in an endurance test program because of the reduced hardness in the scored area and because scoring destroys the profile creating atypical dynamic loads. This problem was corrected by increasing the oil pressure and realigning the oil jets to increase the flow and increase the penetration into the gear mesh. Subsequent running was smoother and quieter and scoring and breakage was eliminated.

The Weibull slopes for the two designs are 1.21 for the standard gear and 1.94 for the HCR design. This indicates that the HCR gears have less scatter than the standard gears tested and that the life comparison with the standard gears is even better for failure rates less 10-percent. Conventional gears, for example are usually designed for a failure rate of .1 percent. The test data reported in Reference 4 on HCR and STD gears represent a Weibull slope of about 1.7 for both configurations. The data reported on herein show slopes a little less for the STD design and a little greater for the HCR design.

The gear patterns on the test gears showed that some teeth appeared to be more heavily loaded than others, suggesting that perhaps tooth spacing or runout could be out of tolerance. The spacing was checked on a number of HCR test gears but was found to be within the specified tolerance. Consideration should be given however, to tightening up the spacing tolerance and runout requirements for HCR gears.

This test shows a high potential for high contact ratio gears in future transmission designs. The designer, however, needs more information than a test of this sort provides. He is less interested in the variability in life at a given load than the variability in an allowable design load or stress at a given no. of cycles, say 10^7 or 10^8; although these two distributions are obviously related.

The test gears used in this program were designed and optimized with the aid of the Hamilton Standard dynamic analysis program. Since variation in tooth thicknesses was precluded by the 1:1 gear ratio, the optimization process concentrated on the amount of tip relief to be applied and the starting point of profile modification. The effectiveness of these tooth modifications is, of course, dependent upon the ability of the computer program to calculate gear tooth compliances and resulting load sharing. If this design tool is to be used in future gear design, it would be helpful to the designers if the predicted gear tooth deflections were checked experimentally and appropriate correction factors established where necessary.

Since most gear design allowables are stress index numbers and not true stresses, and are therefore tagged to the calculation method used; a more accurate design procedure, especially one which calculates dynamic stresses, will require a careful review and updating of the present design allowables to avoid over conservatism.

It should be remembered that, because of the space restrictions imposed by the gear tester, a 1:1 ratio was used in this test program. Additional design options are available to the gear designer when sizing an actual HCR gear set with a higher reduction ratio. These include unequal addenda and top lands as well as unbalanced tooth thicknesses. An additional option is non-symmetrical or buttress teeth which has proved to be beneficial (See Reference 6). It is anticipated that an even greater life increase or weight saving will be possible by exercising these options in an actual gear design.

Metallurgical Evaluation

Metallurgical investigations were conducted on the 6-inch test gear specimens to determine the mode of failure, origin of failure, micro structure of case and core, chemical composition, grain size, grain flow, case depth, and hardness of the case and core. The 6-inch test gear specimens investigated, included two of the standard test gears and two HCRG test gears.

The fractured gear teeth were examined with a low-power stereo microscope to determine the mode and origin of failure. One fractured tooth from each specimen was further examined as follows:

1. The Rockwell hardness was determined for the case and core.
2. Fractured teeth from each process were mounted, etched with 2 percent nital solution, and examined on a metallograph to determine the microstructure of the case and core.
3. The effective case depth was determined by examination of the etched mounts under a Brinell microscope. The effective case depth was determined in terms of "knoop" hardness on a Tukon micro hardness tester for one tooth from each test specimen. The case-core transition point was taken at KHN 542 (approximately equal to R_C 50). The results, presented in Figure 16, have been converted to R_C readings.
4. An investigation of grain size and grain flow was attempted with indiscernible results.

Fracture Analysis

The standard gear S/N B105-00007 evidenced partial fracture of a tooth; Figure 17, views A and B; which was due to a fatigue breakdown of the surface. A small area of spalling extending from a band of damage was found on another tooth, view C. This appears to be the early stage of the condition which led to tooth fracture. S/N B105-00018 contained a single broken tooth; Figure 18, views A and B. Fractographic examination revealed the fracture to be due to fatigue cracking that originated at multiple sites in the root radius, view C. Spalling due to fatigue breakdown, was evidenced on the adjacent tooth, view B. This was the only tooth to exhibit this condition.

The high contact ratio gear S/N B063-00021 exhibited a single tooth fracture that was due to fatigue cracking which originated in the root fillet at the edge of the chamfer; Figure 19, views A-C. S/N B063-0044 did not contain any broken teeth. Scoring and spalling damage similar to that shown in Figure 17 was evident on most teeth.

Metallographic Examination

Two heat treat lots had been used to caburize and harden the parts. Metallographic inspection revealed acceptable case and core microstructures for both lots, shown typically in Figures 20 and 21. Hardness of the case and core and case depths of the test gears are listed in Table 7.

The results of these tests indicate the case hardness to be 1 to 2.5 points low on all gears, and the case depth of gear S/N B105-00021 to be .005 cm (.002 in) below the drawing requirements. Spectrographic analysis indicated the material to be the required 93XX steel. The root radius in all four gears examined conformed to the drawing requirement. Magnetic particle inspection revealed no additional cracks in any gears.

Table 7. Summary of Gear Hardness Case Depth Results.

Tooth Form and Material	Serial No.	Case Hardness R_C	Core Hardness R_C	Effective Case Depth R_C 50 cm (in)
STD-9310	B105-00007	59.0	39.5	.069 (.027)
STD-9310	B105-00018	59.0	40.5	.064 (.025)
HCR-9310	B063-00021	57.5	41.0	.058 (.023)
HCR-9310	B063-00044	58.0	39.5	.064 (.025)
Drawing Requirement		60-64	35-45	.064-.101 (.025-.040)

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1. R.W. Cornell, and W.W. Westervelt: Dynamic Loads and Stressing for High Contact Ratio Spur Gears. Journal of Mechanical Design, Vol 100, January 1978.

2. R.W. Cornell: Compliance and Stress Sensitivity of Spur Gear Teeth, Journal of Mechanical Design, Vol 103, April 1981.

3. K.M. Rosen, and H.K. Frint: Design of High Contact Ratio Gears, Journal of the American Helicopter Society, Vol. 27, No. 4, October 1982.

4. D.P. Townsens, B.B. Baber, and A. Nagy: Evaluation of High-Contact-Ratio Spur Gears with Profile Modification, NASA Technical Paper 1458, September 1979.

5. AGMA Standard 217.01, Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears, 1965.

6. L. Hager, Development of Improved Components for Helicopter Transmissions. USAAVRADCOM Technical Report TR-D-11, November 1984.

CONCLUSIONS

1. The HCR gears scored at a gear bulk temperature of 253°F. The corresponding gear scoring index was calculated to be 460.

2. The 10-percent pitting fatigue life of the HCR gears was approximately two times that of the standard gears at the same load level.

3. The HCR gears exhibited less scatter than the standard gears as reflected by a steeper Weibull curve.

4. Tighter controls should be placed on spacing and runout for HCR gears than for standard gears to derive the most benefit from this design.

5. HCR gears do offer a viable alternative in gear design to reduce weight in a high-quality, high-power transmission.

6. The Hamilton Standard dynamic analysis program is an invaluable tool for the gear designer, particularly for the design of HCR gears.

RECOMMENDATIONS

1. Scoring tests be conducted on the standard gears, as well as, HCR gears to assess relative differences.

2. Additional fatigue tests be conducted on the HCR design to determine design allowables and variability coefficients in bending and pitting.

3. Some attempt be made to assess the effect of tooth spacing accuracy on the life and/or endurance limit values for HCR gears.

4. The compliance of the gear tooth mesh calculated by the HSD dynamic analysis program should be checked experimentally.

5. A review of design allowables applicable to the new design procedure reported on herein should be conducted.

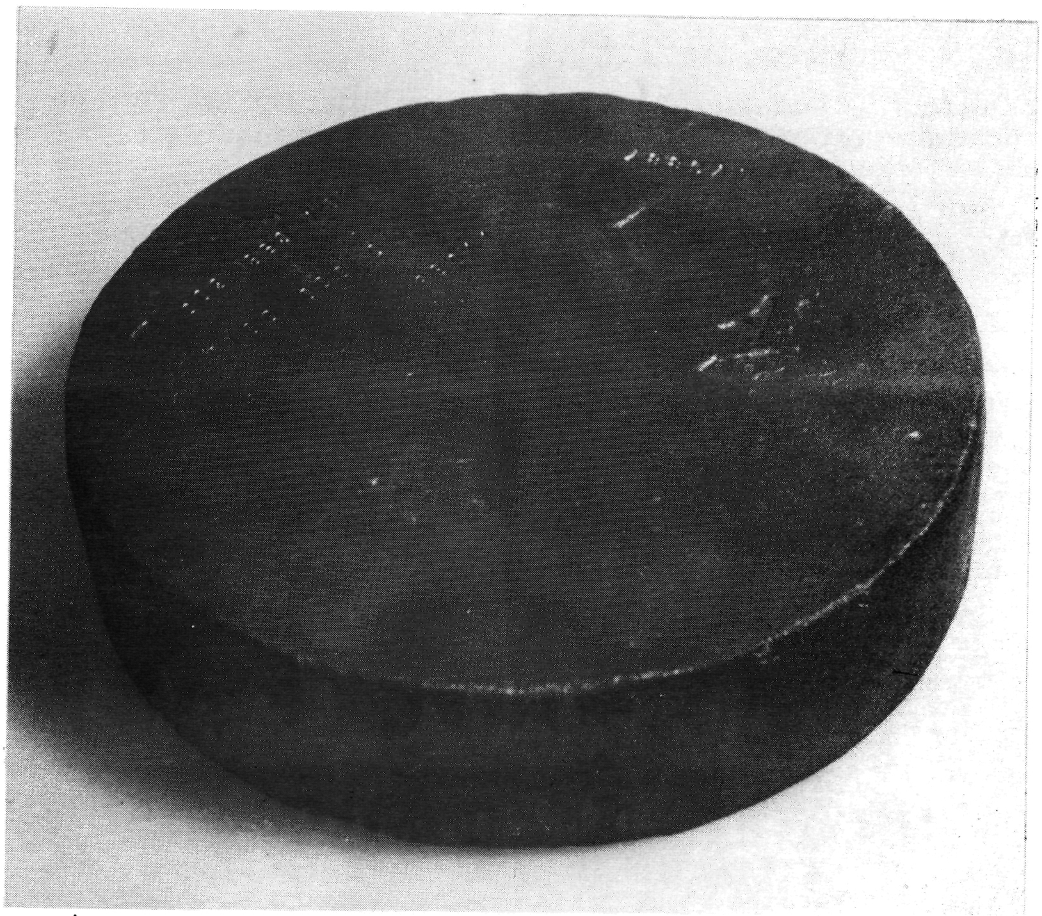
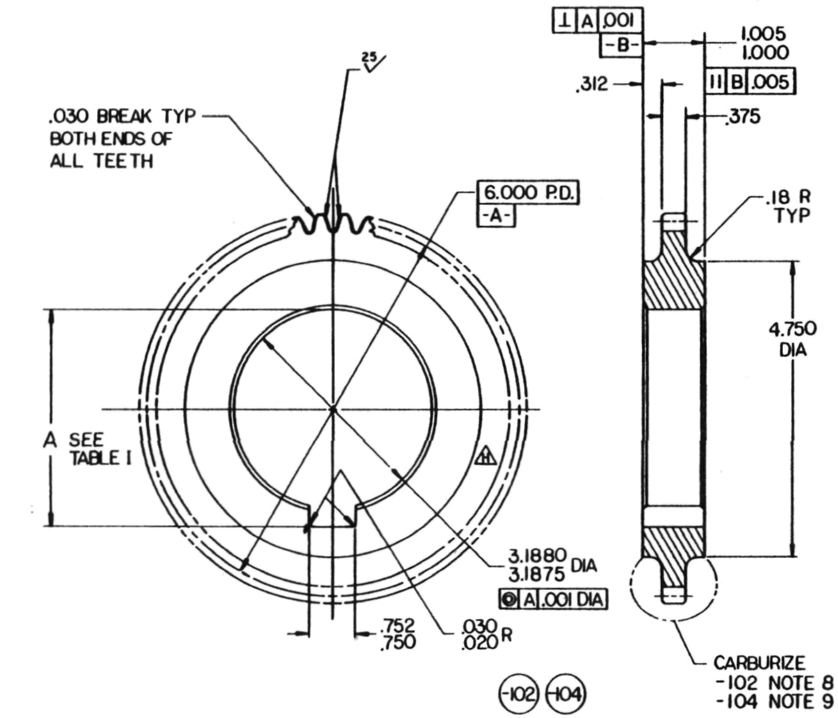


Figure 1. Typical Forging Blank, 6-Inch Test Gears.



EXTERNAL SPUR GEAR DATA

NO. OF TEETH	48
DIAMETRAL PITCH	8
PRESSURE ANGLE	22.50
PITCH DIAMETER	6.0000 -A-
BASE CIRCLE DIAMETER	5.54328
OUTSIDE DIAMETER	6.2450 - 6.2500
ROOT DIAMETER	5.6825 - 5.6925
CHORDAL TOOTH THICKNESS	.1933 - .1943
CHORDAL ADDENDUM	.1241 - .1265
DIMENSION OVER TWO .216 DIA PINS	6.2960 - 6.2983
MAX INVOLUTE PROFILE ERROR AT T.I.F.	-.0000/+.0002
AT TIP	-.0016/-.0020
MAX ACCUMULATED SPACING ERROR BETWEEN ANY TWO TEETH	.0008
MAX TOOTH TO TOOTH SPACING ERROR	.0002
MAX LEAD ERROR	.0002 TOTAL
MAX T.I.F. DIAMETER	5.7996
MATING PART NO.	SAME
ROTATIONAL BACKLASH WITH MATE	.004 - .006
MIN FILLET RADIUS	.0470 - .0570

TABLE I

DASH NO.	A DIM
102	3.350 3.335
104	3.4843 3.4688

NOTES

1 GEARS TO BE MANUFACTURED IN ACCORDANCE WITH SES - 50654, CLASS 2

2 FINISH GRINDING TO BE IN ACCORDANCE WITH SS 8705

3 90 FINISH ALL OVER EXCEPT AS NOTED SS 5100 SHALL APPLY FOR HOLES UNLESS OTHERWISE INDICATED. AREAS OF TRANSITION SHALL CONFORM TO THAT SPECIFIED FOR THE ROUGHEST ADJACENT FINISHED AREA UNLESS OTHERWISE INDICATED

4 UNLESS OTHERWISE NOTED, ALL MACHINED DIAMETERS ON A COMMON CENTERLINE ARE TO BE CONCENTRIC WITHIN .010 DIA

5 BREAK ALL SHARP EDGES .005 - .015 RADIUS UNLESS OTHERWISE SPECIFIED

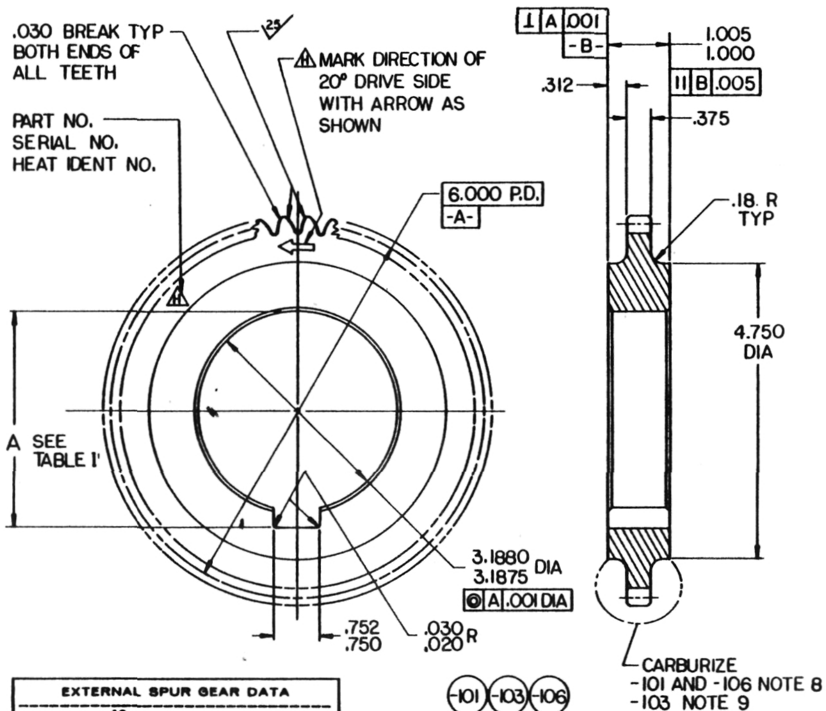
6 SERIALIZE PER SS 9070 USING ITEM IDENTIFICATION PREFIX

7 HEAT IDENTIFICATION MARKING SHALL BE TRANSFERRED FROM THE FORGING TO THE MACHINED PART

8 CARBURIZE AREAS INDICATED TO PRODUCE .025-.040 DEPTH OF CASE IN THE FINISHED PART. BEFORE FINISH MACHINING HEAT TREAT TO ROCKWELL C 60-64 OR EQUIVALENT CASE HARDNESS WITH ROCKWELL C 35-45 CORE HARDNESS. CARBURIZE AND HEAT TREAT PER SS8015 WITH THE FOLLOWING EXCEPTIONS: CARBURIZE AT 1700 - 1750 °F, OIL QUENCH FROM CARBURIZING TEMPERATURE FOLLOWED BY A CONDITIONING AT 1100-1300°F. AUSTENITIZE AT 1610-1660 °F, OIL QUENCH, DEEP FREEZE AND DOUBLE TEMPER AT 590 - 610 °F FOR 2 + 2 HOURS

9 CARBURIZE AREAS INDICATED PER SS 8015 TO PRODUCE .025 - .040 DEPTH OF CASE IN THE FINISHED PART. BEFORE FINISH MACHINING HEAT TREAT TO ROCKWELL C 60-64 OR EQUIVALENT CASE HARDNESS WITH ROCKWELL C 35 - 45 CORE HARDNESS PER SS 8015

Figure 2. Six-Inch Test Gear, Standard Tooth Form.



EXTERNAL SPUR GEAR DATA

NO. OF TEETH	48
DIAMETRAL PITCH	8
PRESSURE ANGLE	20°(DRIVE)
PITCH DIAMETER	6.000 -A-
BASE CIRCLE DIAMETER	5.63815 / 5.43785
OUTSIDE DIAMETER	6.3394 – 6.3444
ROOT DIAMETER	5.6057 – 5.6157
CHORDAL TOOTH THICKNESS	.1933 – .1943
CHORDAL ADDENDUM	.1713 – .1737
DIMENSION OVER TWO .2880 DIA PINS	6.5331 - 6.5351
MAX INVOLUTE PROFILE ERROR	
DRIVE SIDE	SEE TABLE III
COAST SIDE AT T.I.F.	-.0000/+.0003 AT TIP -.0000/+.0004
MAX ACCUMULATED SPACING ERROR BETWEEN ANY TWO TEETH	.0008
MAX TOOTH TO TOOTH SPACING ERROR	.0002
MAX LEAD ERROR (PER INCH OF FACE)	.0002 TOTAL
MAX T.I.F. DIAMETER	5.7634
MATING PART NO.	SAME
ROTATIONAL BACKLASH WITH MATE	.004 - .006
MIN FILLET RADIUS	.0337 - .0437

TABLE II

DASH NO.	A DIM
101 & 106	3.350 3.335
103	3.4843 3.4688

NOTES

1 GEARS TO BE MANUFACTURED IN ACCORDANCE WITH SES-50654, CLASS 2

2 FINISH GRINDING TO BE IN ACCORDANCE WITH SS8705

3 20 FINISH ALL OVER EXCEPT AS NOTED SS5100 SHALL APPLY FOR HOLES UNLESS OTHERWISE INDICATED. AREAS OF TRANSITION SHALL CONFORM TO THAT SPECIFIED FOR THE ROUGHEST ADJACENT FINISHED AREA UNLESS OTHERWISE INDICATED

4 UNLESS OTHERWISE NOTED, ALL MACHINED DIAMETERS ON A COMMON CENTERLINE ARE TO BE CONCENTRIC WITHIN .010 DIA

5 BREAK ALL SHARP EDGES .005 - .015 RADIUS UNLESS OTHERWISE SPECIFIED

6 SERIALIZE PER SS9070 USING ITEM IDENTIFICATION PREFIX

7 HEAT IDENTIFICATION MARKING SHALL BE TRANSFERRED FROM THE FORGING TO THE MACHINED PART

8 CARBURIZE AREAS INDICATED TO PRODUCE .025-.040 DEPTH OF CASE IN THE FINISHED PART. BEFORE FINISH MACHINING HEAT TREAT TO ROCKWELL C60-64 OR EQUIVALENT CASE HARDNESS WITH ROCKWELL C35-45 CORE HARDNESS. CARBURIZE AND HEAT TREAT PER SS8015 WITH THE FOLLOWING EXCEPTIONS: CARBURIZE AT 1700-1750 °F, OIL QUENCH FROM CARBURIZING TEMPERATURE FOLLOWED BY A CONDITIONING AT 1100-1300 °F. AUSTENITIZE AT 1610-1660 °F, OIL QUENCH, DEEP FREEZE AND DOUBLE TEMPER AT 590 - 610 °F FOR 2 + 2 HOURS

9 CARBURIZE AREAS INDICATED PER SS8015 TO PRODUCE .025-.040 DEPTH OF CASE IN THE FINISHED PART. BEFORE FINISH MACHINING HEAT TREAT TO ROCKWELL C60-64 OR EQUIVALENT CASE HARDNESS WITH ROCKWELL C35-45 CORE HARDNESS PER SS8015

Figure 3. Six-Inch Test Gear, HCRB Tooth Form

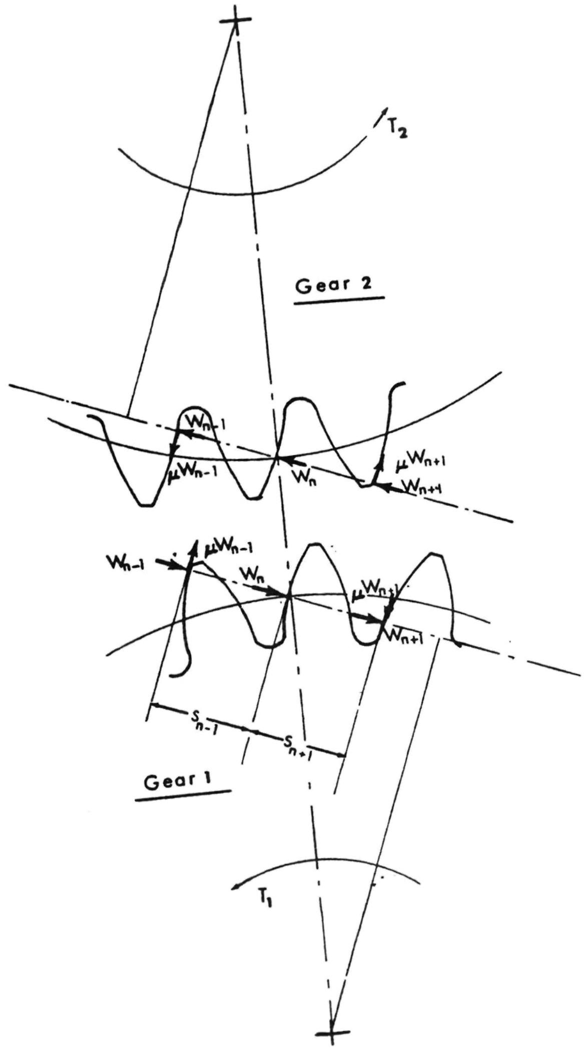


Figure 4. Free Body Diagram of Mesh, High Contact Ratio Gears

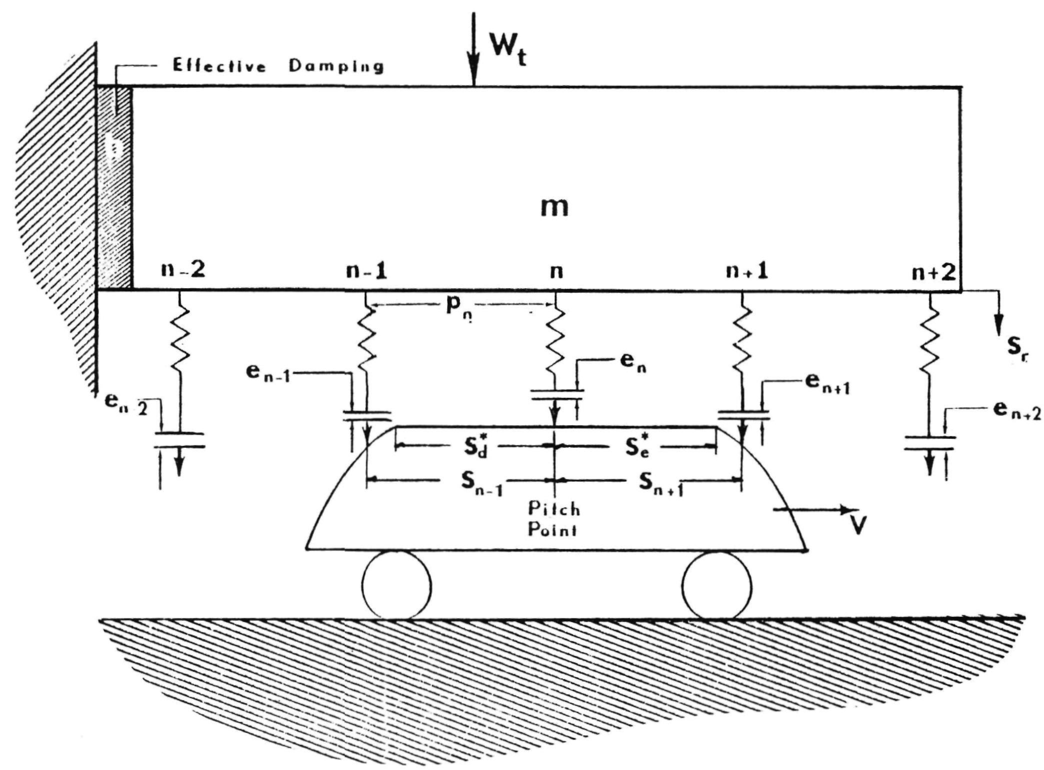


Figure 5. High Contact Ratio Dynamic Load Model

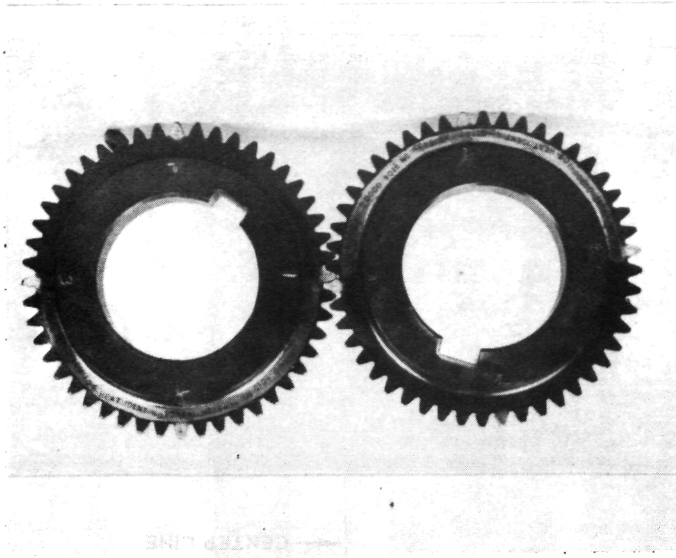


Figure 6. Fabricated 6-Inch Test Gears,
Standard Tooth Form.

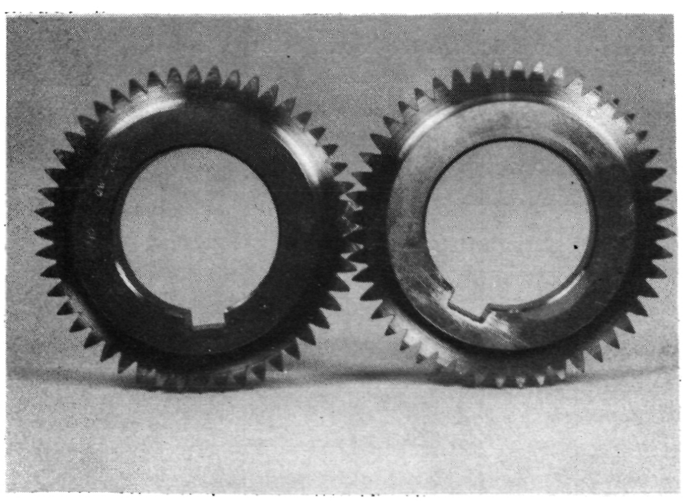


Figure 7. Fabricated 6-Inch Test Gears,
HCR Tooth Form.

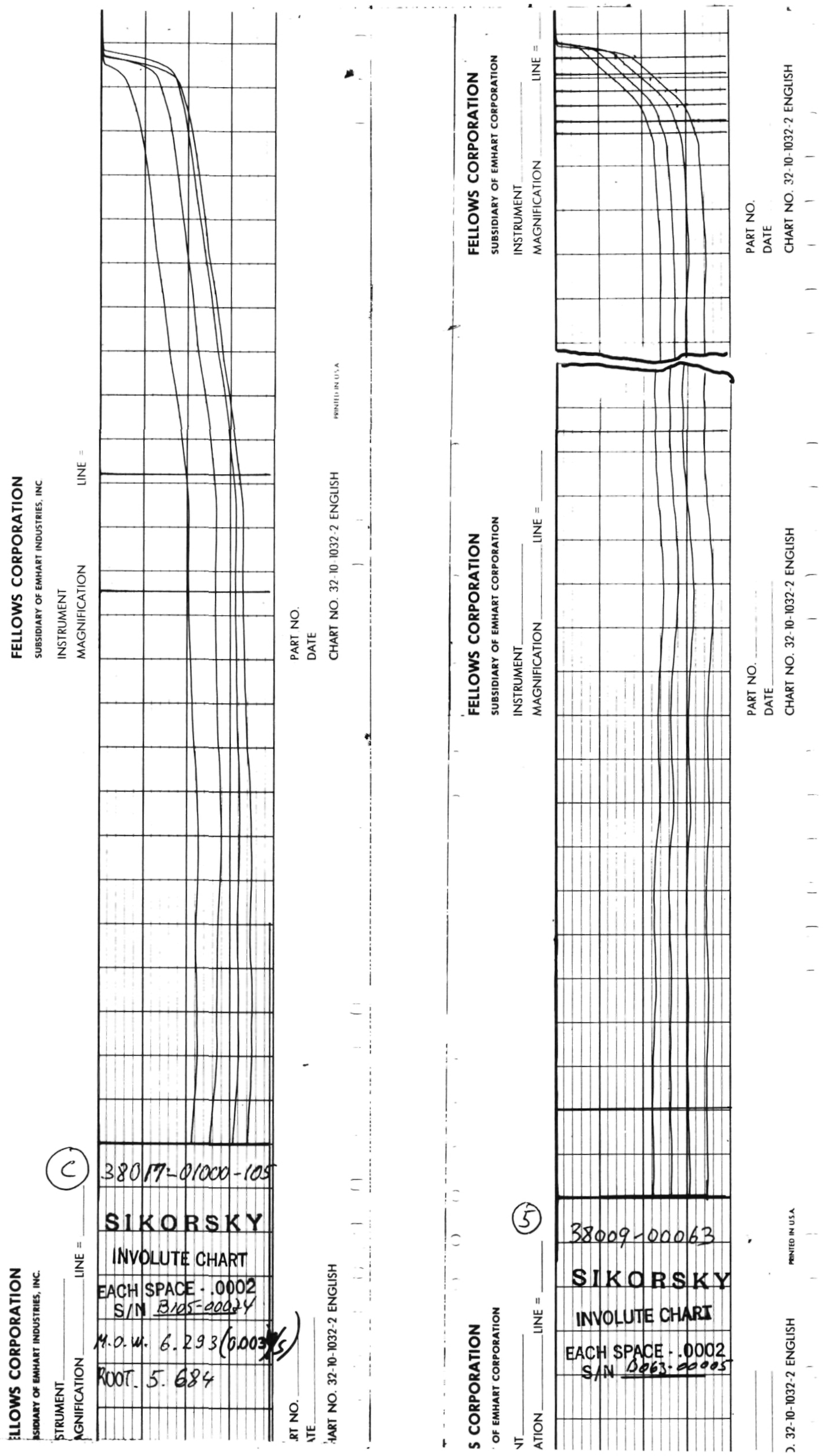


Figure 8. Involute Profile Traces For Test Gears

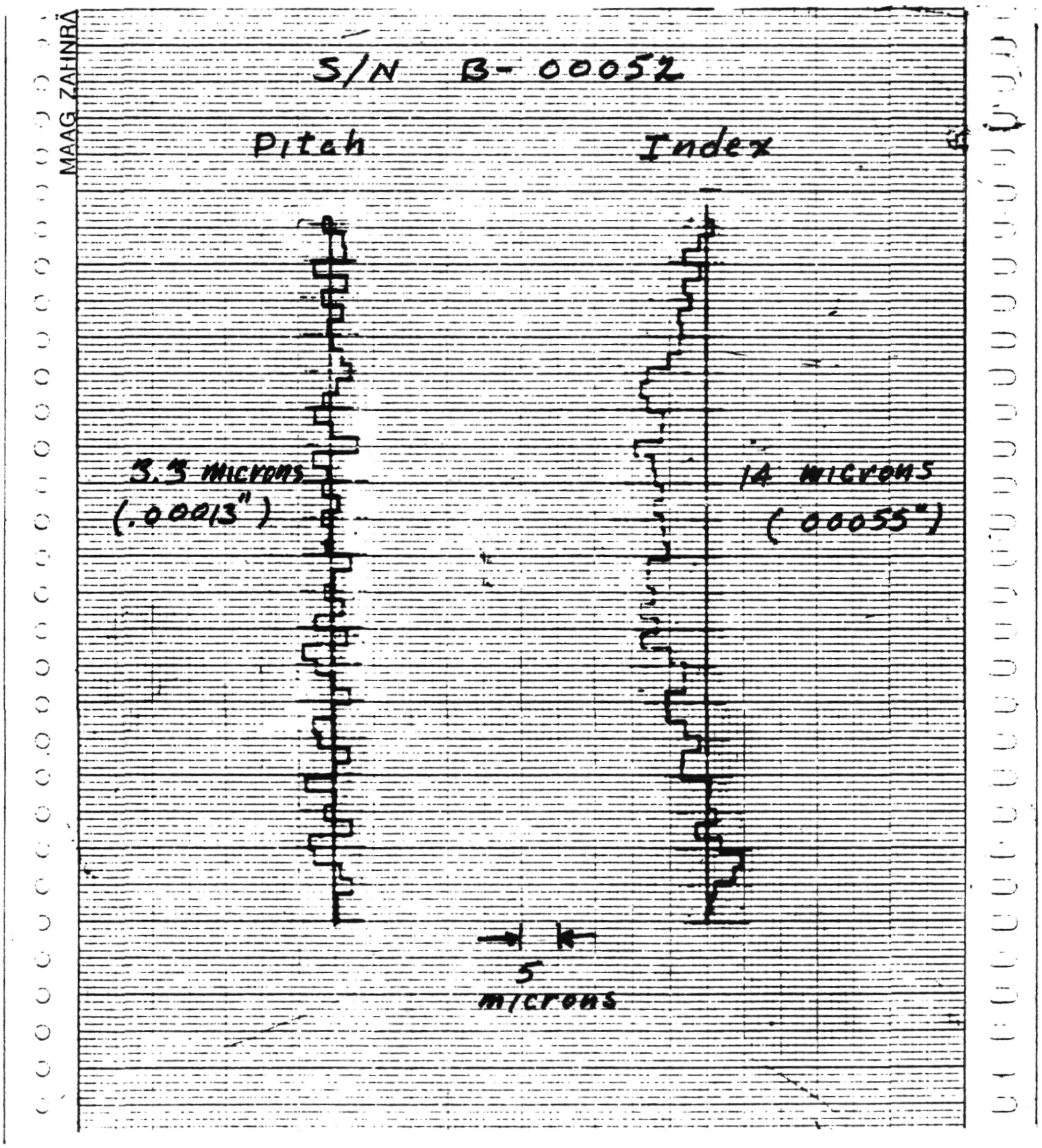


Figure 9. Test Gear Spacing and Index Errors

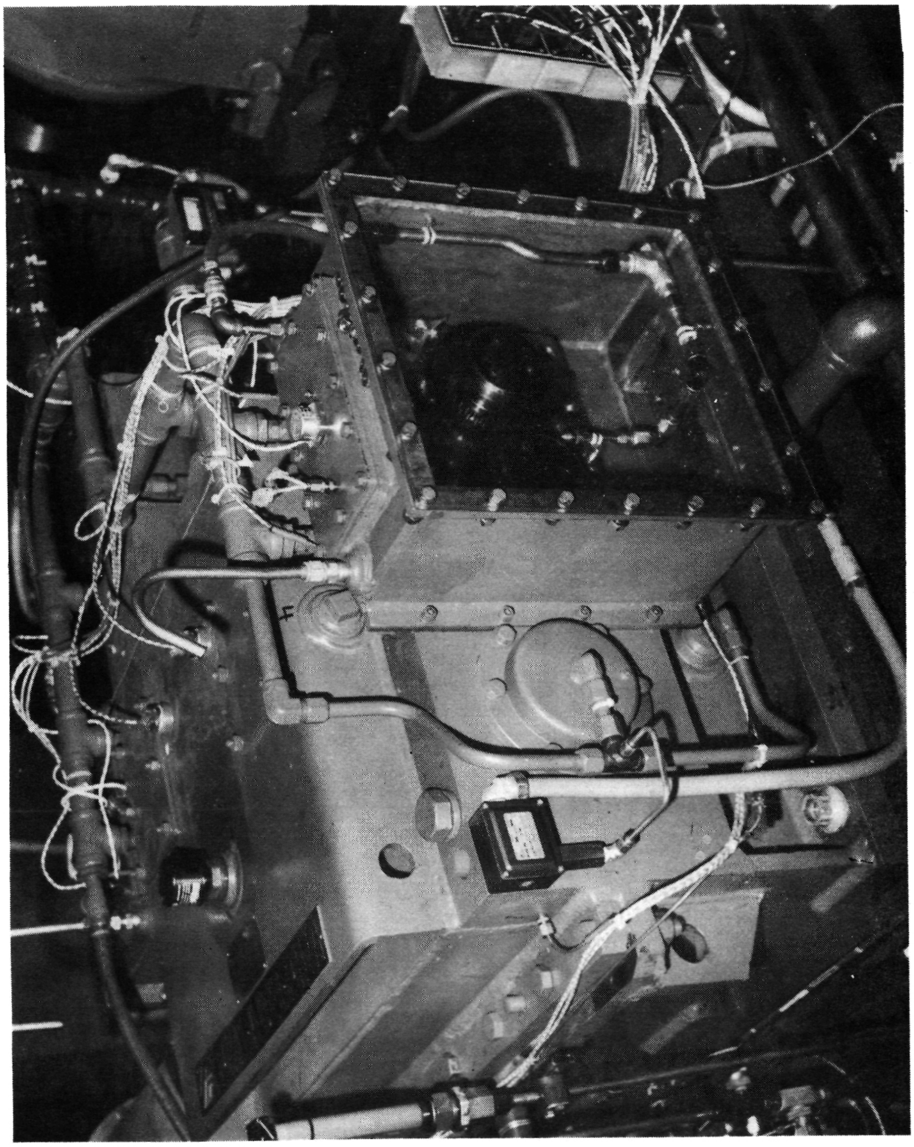


Figure 10. Six-Inch Gear Test Facility, Side Number 1.

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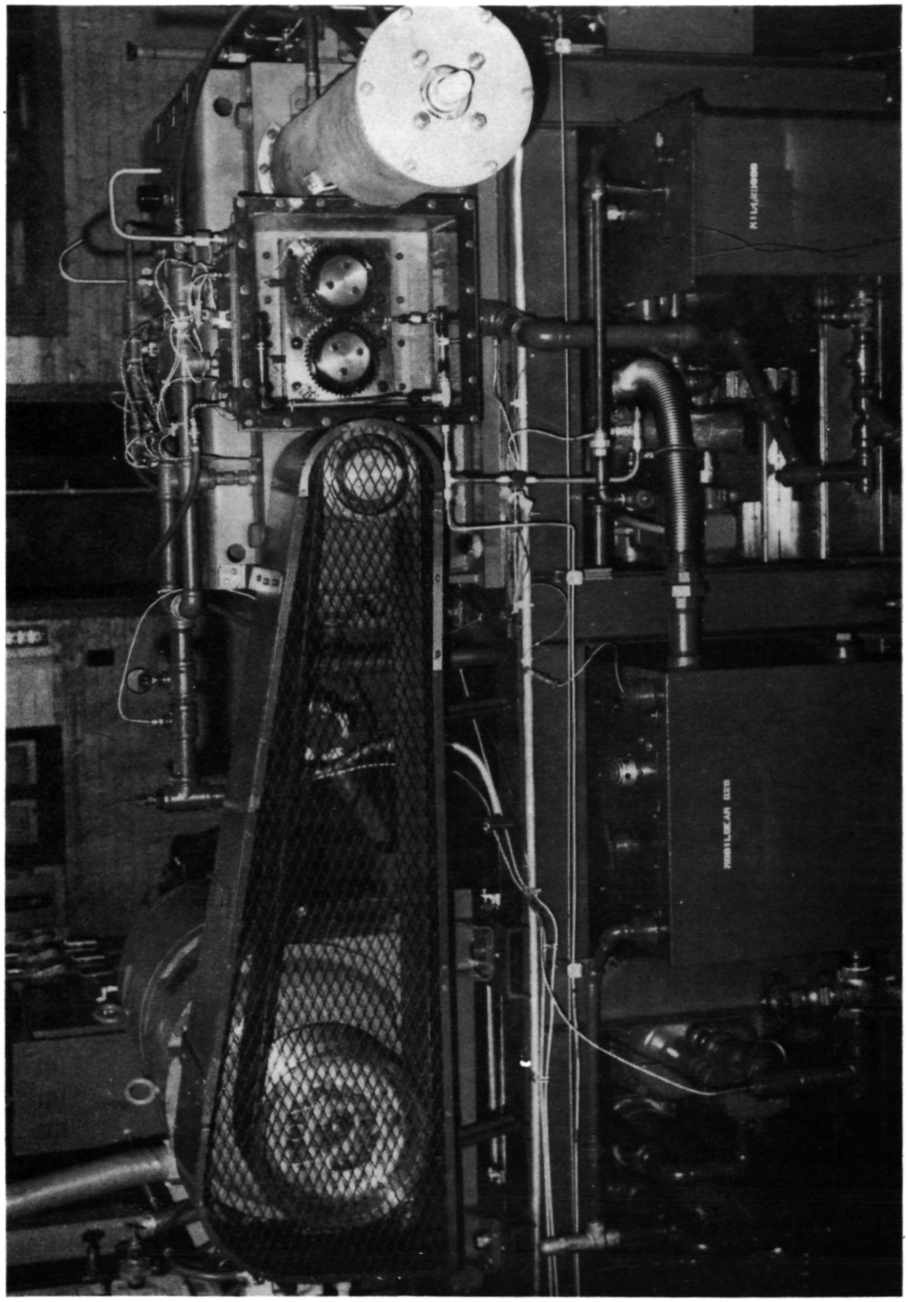


Figure 11. Six-Inch Gear Test Facility, Test Side Number 2.

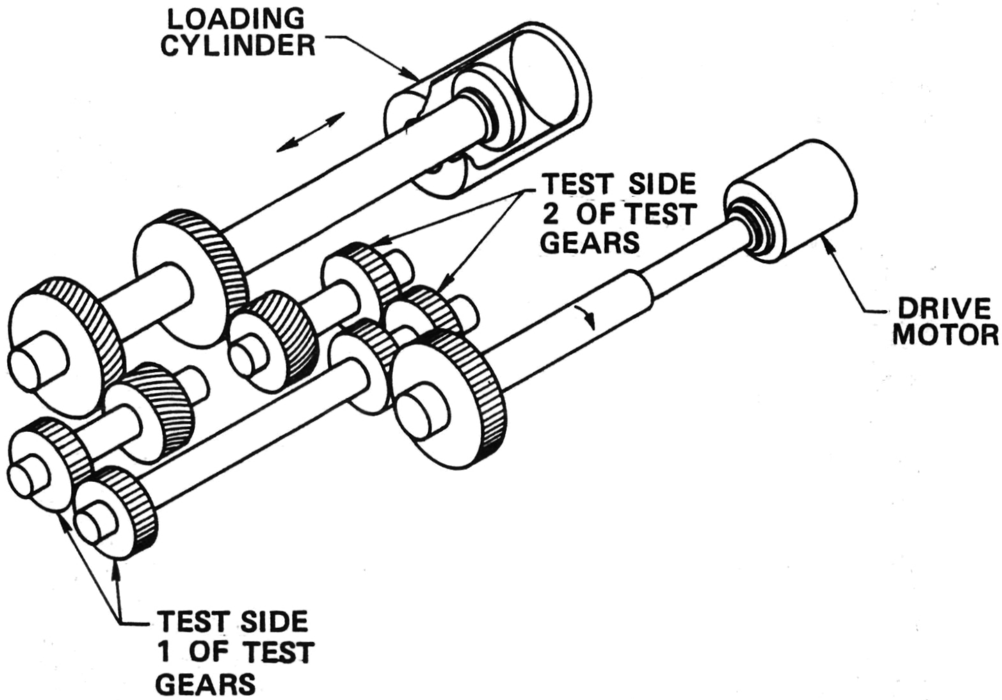


Figure 12. Schematic, Sikorsky 6-Inch Gear Tester.

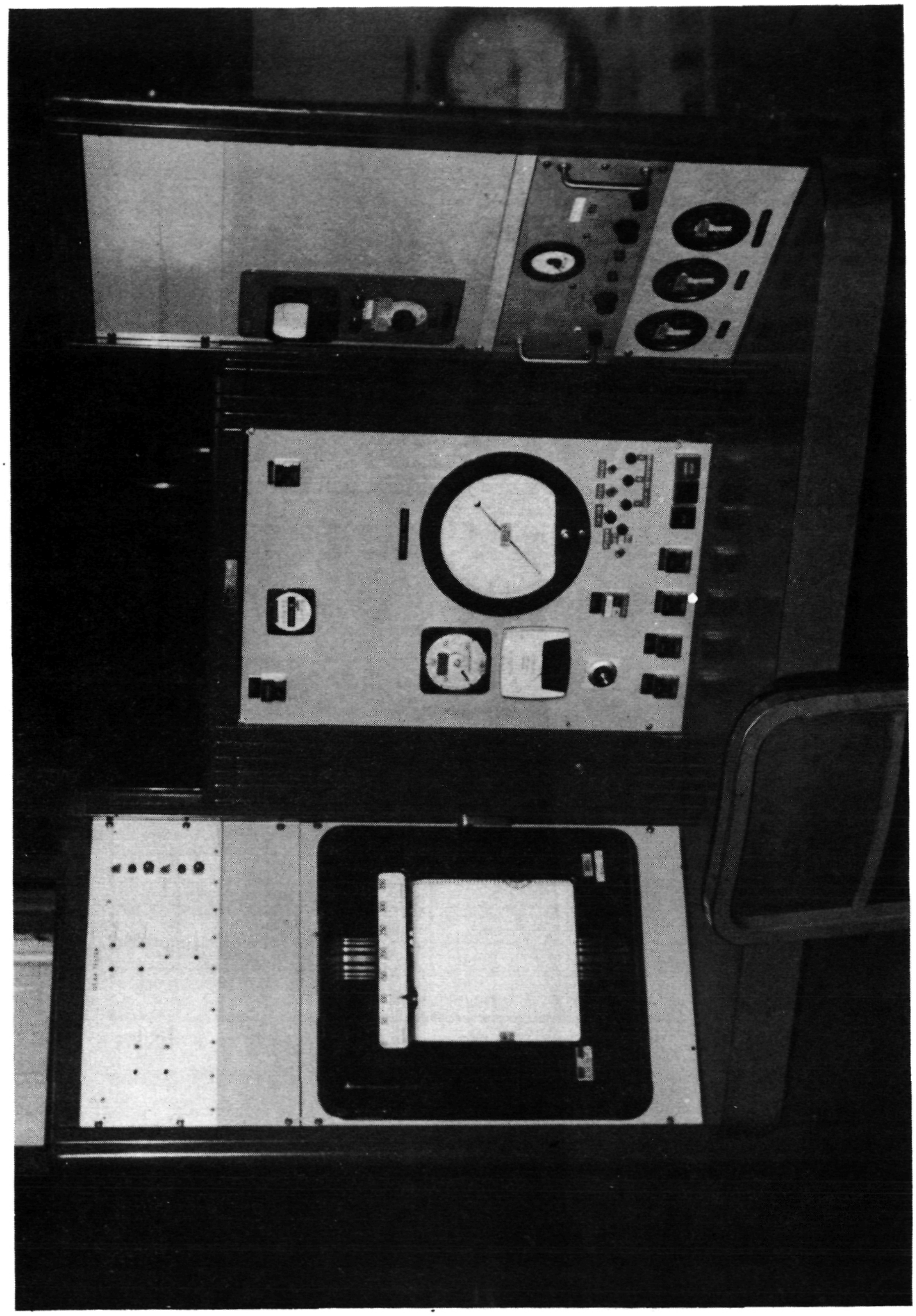


Figure 13. Operator's Console, 6-Inch Gear Test Facility.

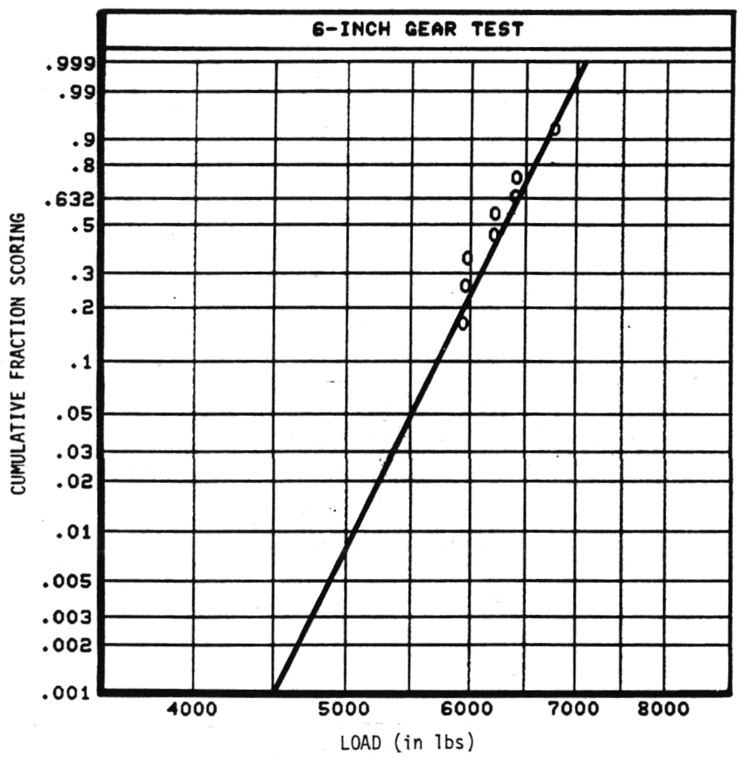
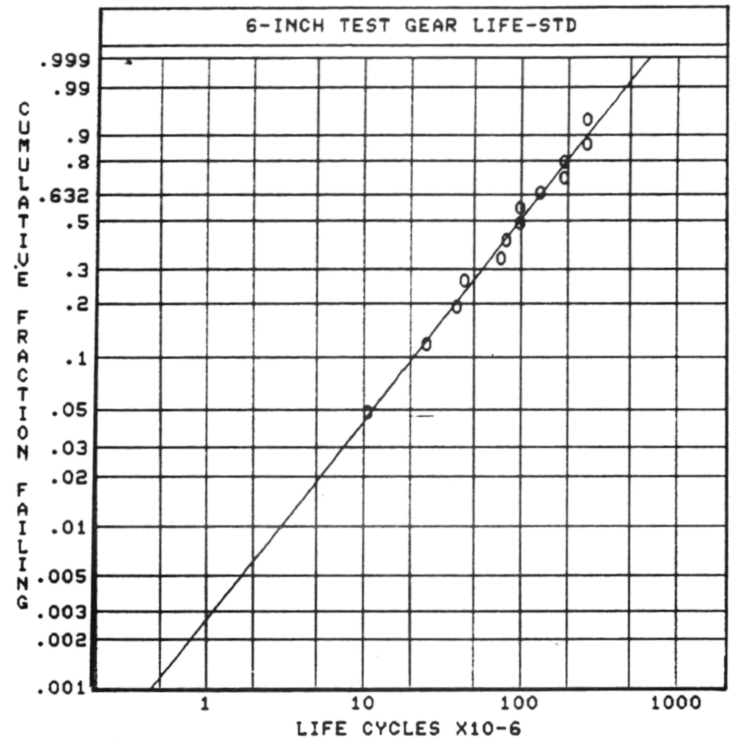


Figure 14. HCR Gear Scoring Test Results



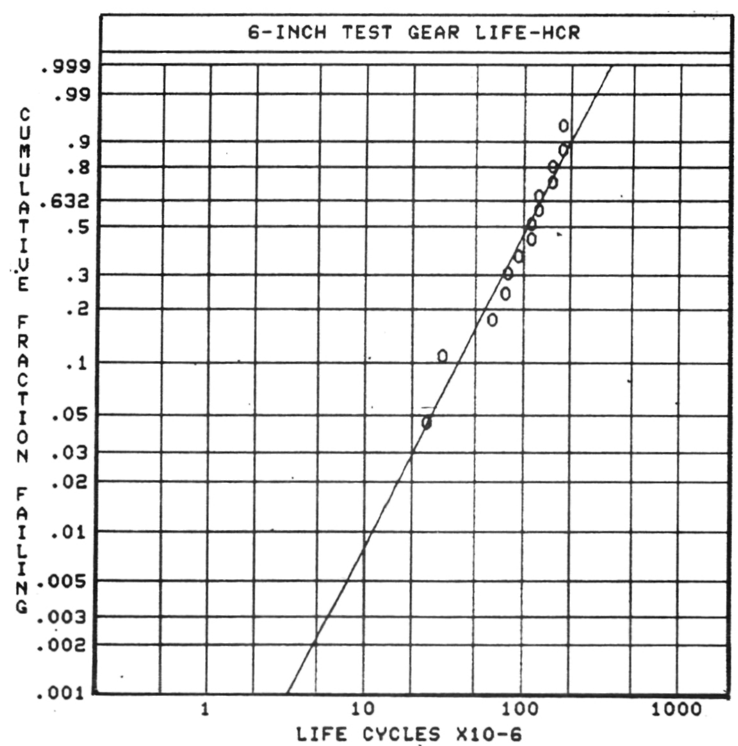
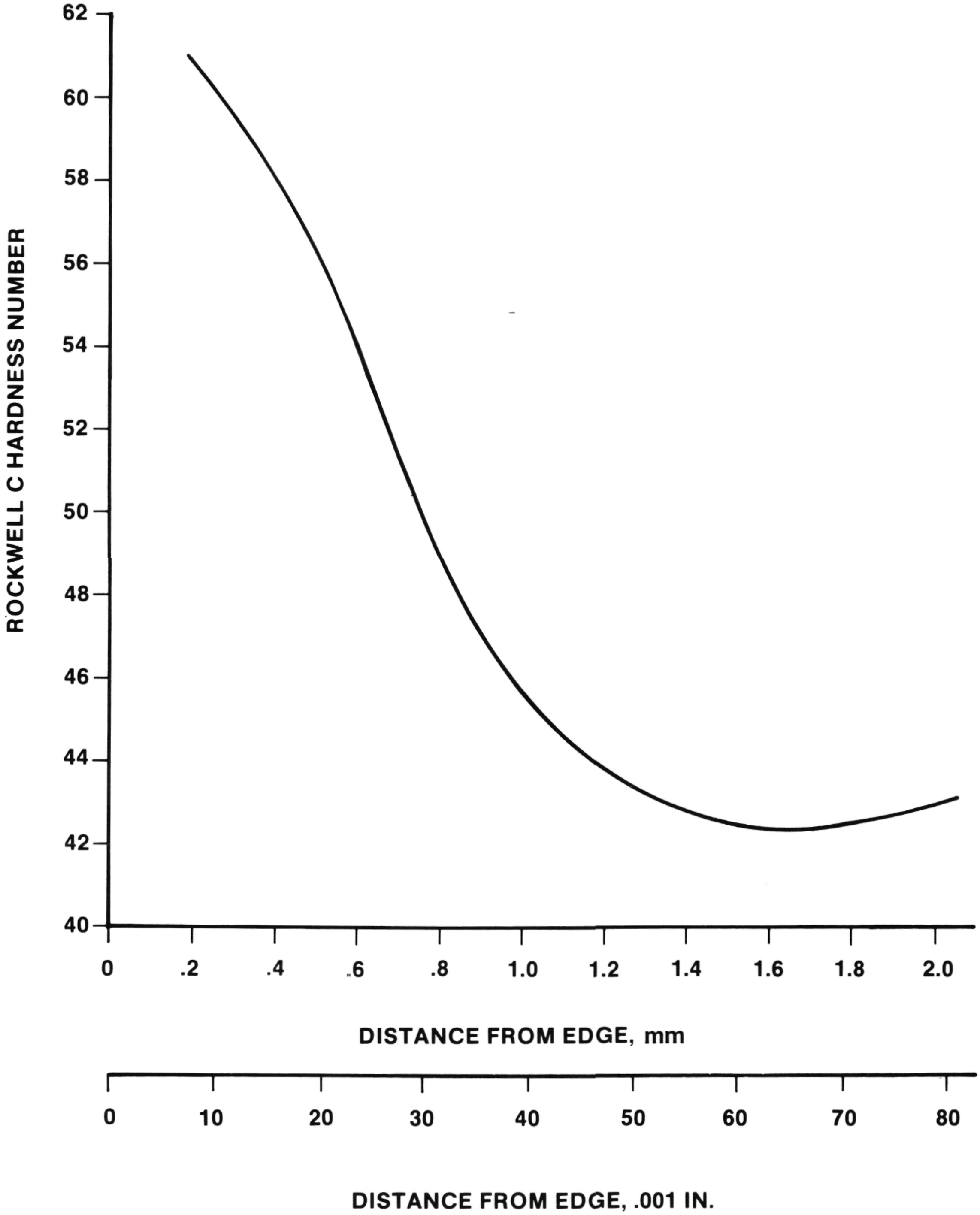
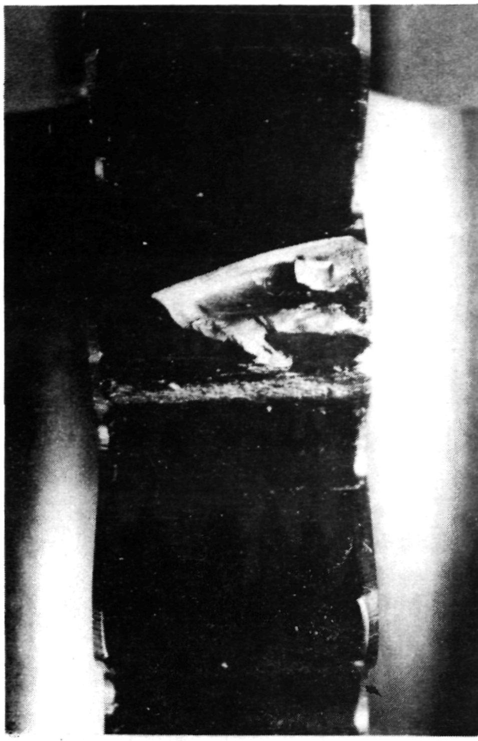
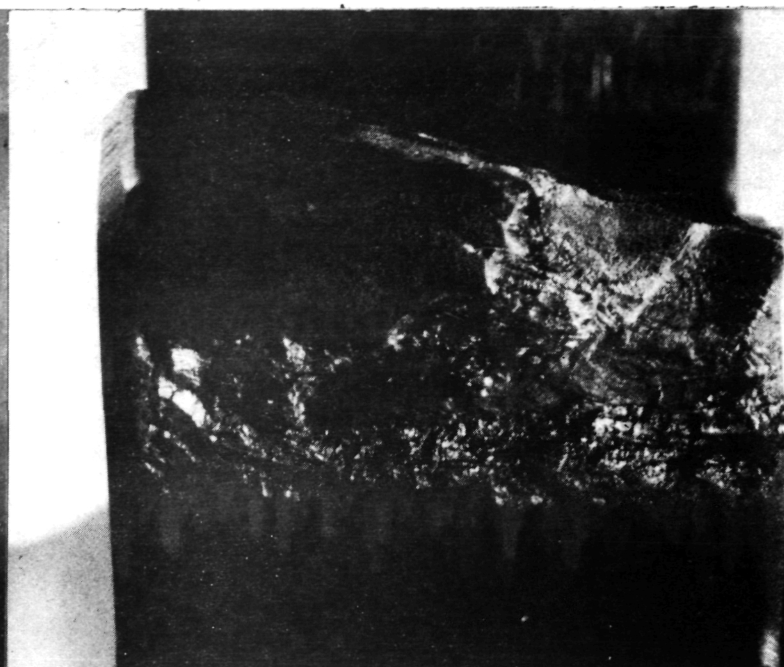


Figure 15. HCR and Standard Gear Pitting Test Results



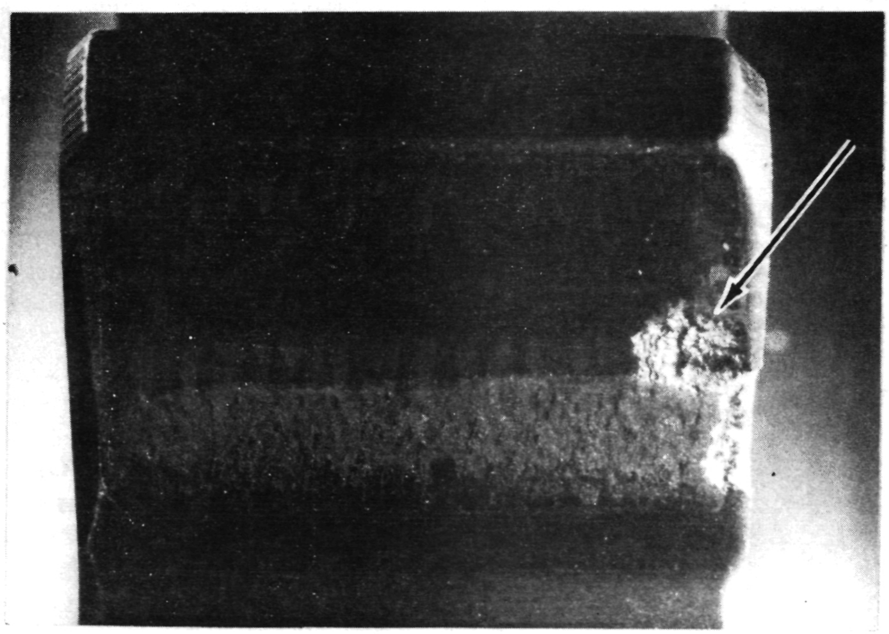
Figures 16. Test Gear Microhardness Data





View B Spalled surface from which fracture extended.

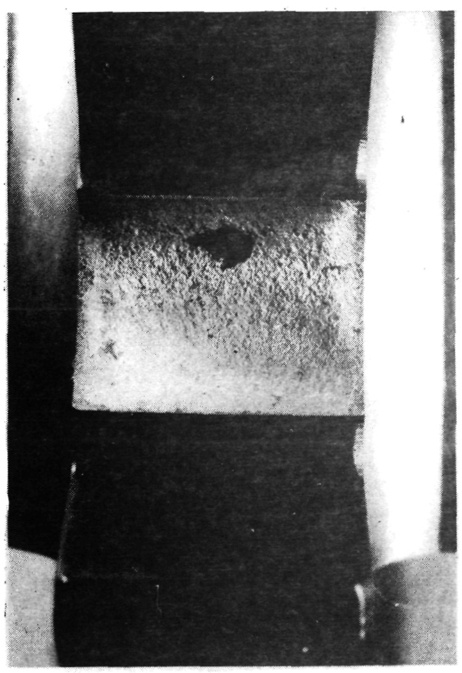
A Configuration of fractured tooth on gear S/N B105-00007.



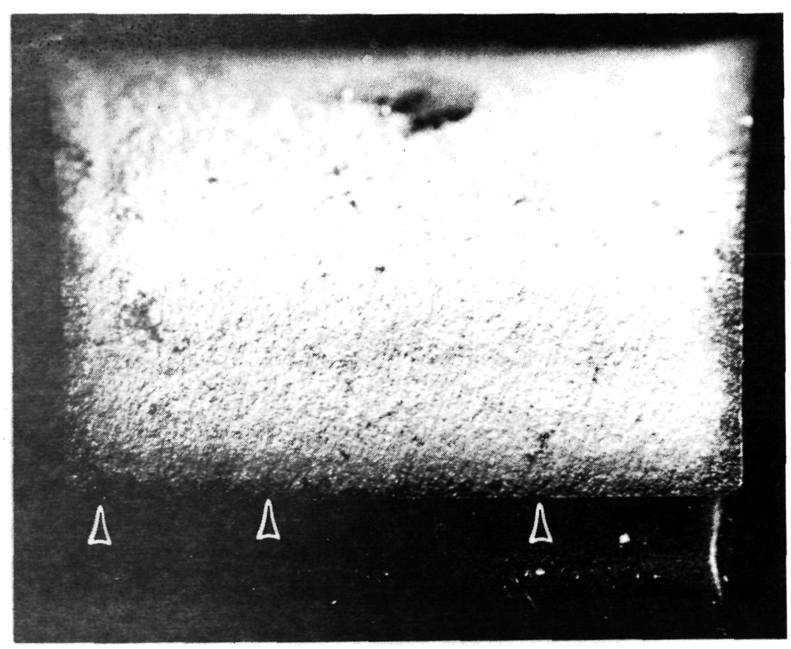
View C Scoring and early stage of spalling. 8X

Figure 17. Test Gear Failures, S/N B105-00007

View A 38017-01000-105
Standard Configuration
S/N B105-00018



View B 4X
Fracture surface indicative of fatigue. Note: spalling of adjacent tooth.

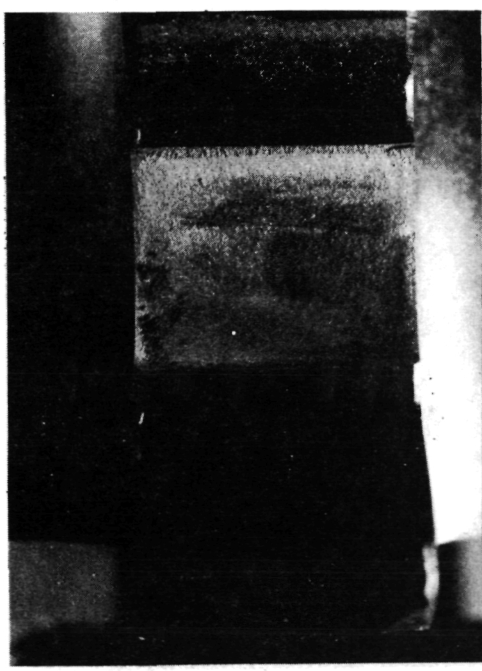


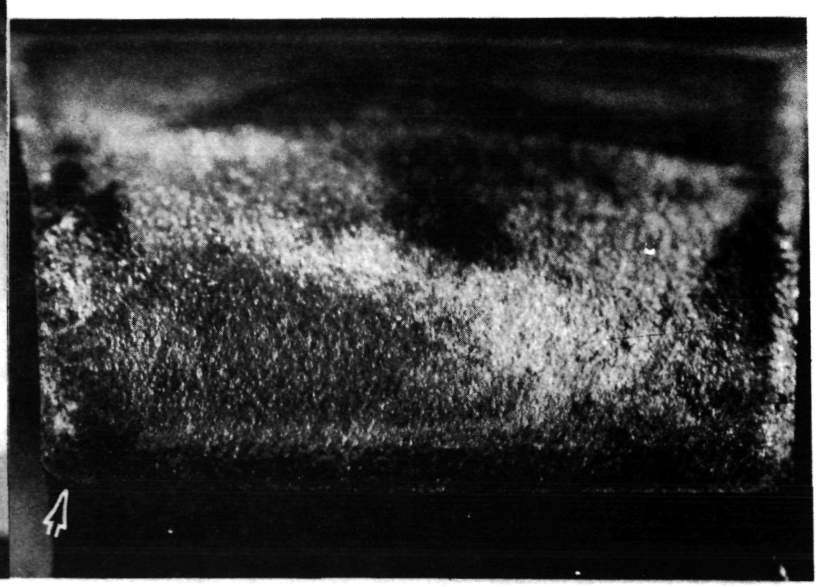
View C

Multiple origins along filet radius.

Figure 18. Test Gear Failures, S/N B105-00018

View A H.C.R. Gear 38009-00063
S/N B063-00021.





View B Fracture Surface View C Fatigue origin at edge of chamfer. 1

Figure 19. Test Gear Failures, S/N B063-00021

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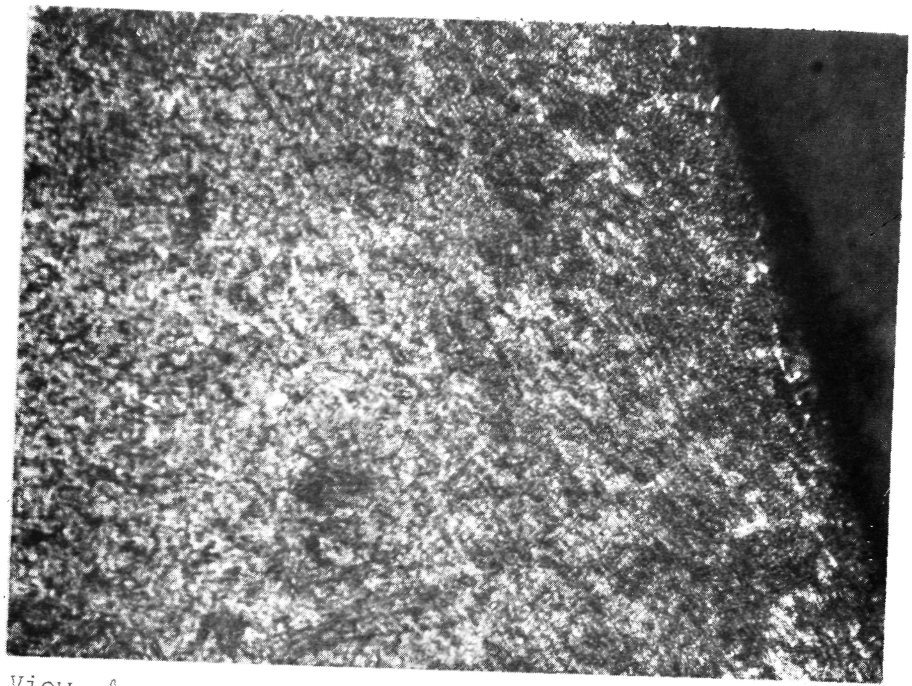
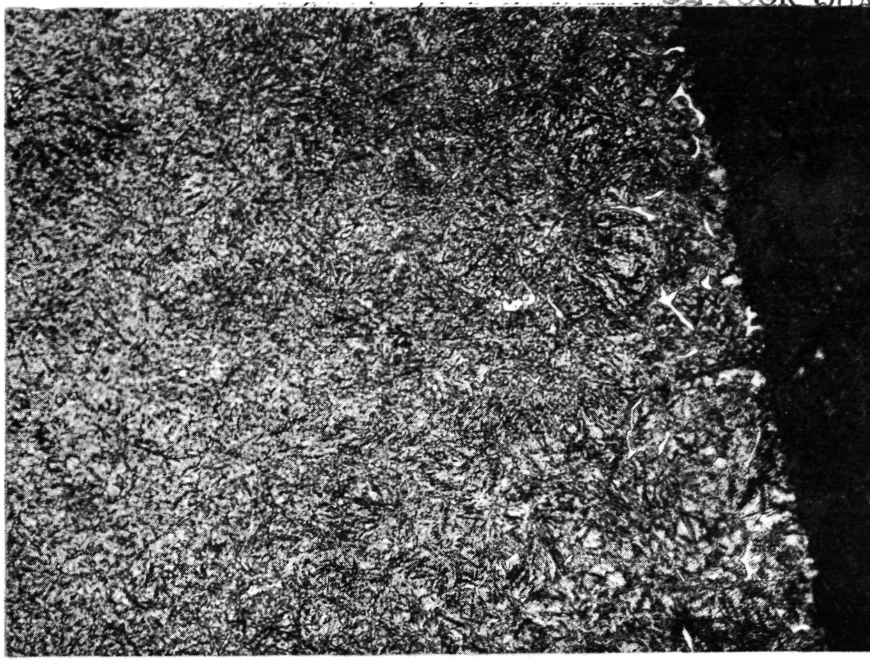


View A Case microstructure. 500X

View B Core microstructure. Gear S/N 44 - typical for H.T. 6662.

Figure 20. Test Gear Microstructure, S/N B063-00044



View A Case microstructure 500X

View B Core microstructure. Gear S/N B105-00018.
Typical of heat-treat 6631.

Figure 21. Test Gear Microstructure, S/N B105-00018